Observations on and potential trends for mechanically supercharging a downsized passenger car engine – A review

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Abstract:

Engine downsizing is a proven approach to achieve superior fuel efficiency. It is conventionally achieved by reducing the engine swept volume and employing some means of increasing specific output to achieve the desired installed engine power, usually in the form of an exhaust-driven turbocharger. However due to the perceptible time needed for the turbocharger system to generate the required boost pressure, turbocharged engines characteristic degraded driveability when compared to their naturally-aspirated counterparts. Mechanical supercharging refers to the technology that compresses the intake air using the energy taken directly from the engine crankshaft. It is anticipated that engine downsizing that is realized either solely by a supercharger or by a combination of supercharger and turbocharger would enhance a vehicle's driveability without significantly compromising the fuel consumption at an engine level compared to the downsizing by turbocharging. The capability of the supercharger system to eliminate the large exhaust back pressure, reduce the pulsation interference and mitigate the surge issue of a turbocharged engine in a compound charging system will offset some of the fuel consumption penalty incurred in driving the supercharger. This, combined with an optimized down-speeding strategy, might further improve a downsized engine's fuel efficiency performance whilst still

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enhancing its driveability and performance at a vehicle level.

This paper will first review the fundamentals and types of supercharger that are currently or have been used in passenger car engines. Next, the relationships between downsizing, driveability and down-speeding are introduced to identify the improved synergies between the engine and the boosting machine. Then, mass production and prototype downsized supercharged passenger car engines are briefly described, followed by a detailed review of the current state-of-the-art supercharging technologies that are in production as opposed to the approaches that are currently only being investigated at a research level. Finally, the trends for mechanically supercharging a passenger car engine are discussed, with the aim of identifying potential development directions for the future. Low-end torque enhancement, transient driveability improvement and low load parasitic loss reduction are the three main development directions for a supercharger system, among which the adoption of a CVT to decouple the supercharger speed from the engine speed, compressor isentropic and volumetric efficiency improvement and supercharger mechanism innovation seem to be the potential trends for mechanically supercharging a passenger car engine.

Keywords: **Mechanically**, **Supercharging**, **Downsizing**, **Passenger car**, **Engine**

1 Introduction

Engine downsizing, which is the use of a smaller-swept-volume engine to provide the power of a larger one, is widely accepted as one of the most viable solutions to address the fuel economy and environment issues facing passenger car engines [1-7]. Reduced pumping loss, improved heat transfer and better friction condition, thus shifting the engine operating points into a more efficient area, are the major reasons for improved fuel efficiency in the frequently-used areas of low-load engine operation. The rated power and torque are conventionally recovered by means of turbocharging and supercharging [8-9]. Turbocharged engines are generally more fuel-efficient as they utilize a proportion of otherwise-wasted exhaust gas energy. But most turbocharged engines are not 'fun' enough due to the perceptible time needed for the turbocharging system to generate the required boost (so-called 'turbo-lag'). Supercharged engines on the other hand do not suffer from this problem, because the compressor of a supercharged engine is directly driven by the engine crankshaft; however, they are not as efficient as their turbocharged counterparts and increase engine friction directly.

Although most of the downsized passenger car engines in production are turbocharged and the driving performance of turbocharged vehicles has been greatly improved by the development of turbocharger system itself (for example: reduction of turbocharger inertia) and proper charging system matching, with the requirement of further downsizing to achieve superior fuel efficiency [10] to satisfy ever-more strict fuel consumption regulations, supercharging technology may have to be introduced to increase the low-end torque and improve the transient performance when needed.

It might be worth noting that supercharging a passenger car engine can also address some other inherent issues of a turbocharged alternative, among which the elimination of pulsation interference and the capability to reduce the high exhaust back pressure are the two major aspects. Improvement of the turbocharging system itself, such as modifying a conventional turbine to a twin-entry one [11,12] to facilitate scavenging and improve the low-end torque particularly in four cylinder groups [13] and/or employing a variable geometry turbocharger (VGT) to achieve optimum efficiency in a wider range of speeds and loads whilst improving the transient performance [14-15] and/or adopting divided exhaust period (DEP) [16-27] or the so-called turbo-discharging concept [28-30] to reduce the backpressure to further improve a turbocharged engine's performance, can achieve some benefits but not as significantly when compared to a supercharged counterpart.

Mechanically supercharging an engine was commonplace in aero-engines [31-32], but the development of this technology for a passenger car only appears in recent years. This is especially the case for a compound-charging arrangement, in which two or more charging devices are used to provide more system capability [33-36]. Considering the strong possibility that the supercharging technology will be applied to more passenger car engines in the near future to facilitate further downsizing and more aggressive down-speeding, either in a Supercharger-Turbocharger or Turbocharger-Supercharger arrangement, a review of the current and concept approach to achieving supercharging is necessary.

In this paper, fundamentals and types of supercharger will first be briefly introduced, following which the paper will describe the relationship between downsizing, driveability and down-speeding for the purpose of understanding the interactions between fuel economy and vehicle performance. Some current mass production and highly downsized prototype passenger car engine will then be discussed. Finally, conventional and novel approaches to supercharging are compared, with the aim of identifying potential developments in the future.

Modeling methodologies that can capture the dynamic response of different boosting systems are detailed in a separate section, as it is believed that only proper modelling is able to assist in evaluating the engine's fuel efficiency performance with different transient behavior in a typical driving cycle.

2 Fundamentals and types of supercharger

The term 'supercharger' in this paper refers to an air pump that increases the pressure and thus density of the charge air supplied to an internal combustion engine by directly connecting to an engine's crankshaft (and thus taking power from it) by means of a belt, gear or chain [37].

In general, there are two main types of superchargers defined according to the method of gas transfer: Positive displacement and centrifugal compressors [37].

Positive displacement units offer a relatively constant boost characteristic since they pump air at a fixed rate relative to the engine speed and supercharger size. In many respects, a positive-displacement supercharger may be a more easily integrated device than an aerodynamic one. It is a lower speed machine and can therefore have a simple drive system to the crankshaft and it also has air consumption characteristics similar to a typical internal combustion engine [38].

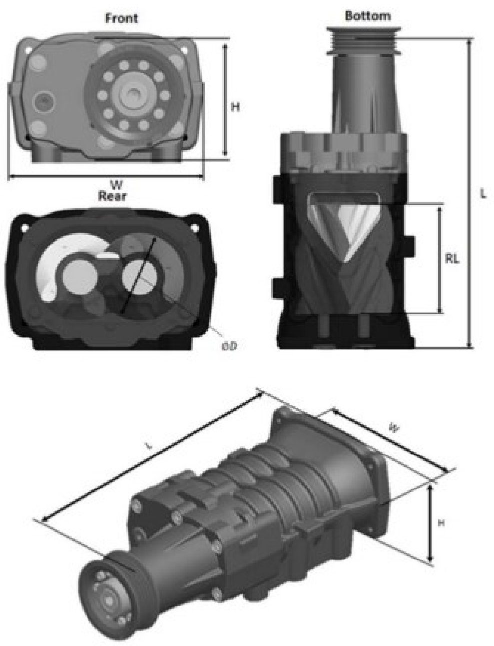

Figure 1. Eaton TVS R-Series supercharger view. [44]

Figure 2. Lysholm screw supercharger. [45]

For passenger car engines, the most frequently used positive displacement superchargers are the Roots[2] and twin-screw types (the latter incorporating internal compression and therefore correctly termed a compressor, the former achieving all compression externally and therefore being properly a 'blower') [2,39]. Both Volkswagen and Volvo used Eaton (Roots-type) superchargers in their production compound-charged gasoline engines [34-35,40], and Lotus, Audi and Jaguar have used them as single-stage boosting devices in recent production engines [41-43].

Figures 1 and 2 show an Eaton TVS Roots-type supercharger [44] and the lobes from a Lysholm screw-type supercharger [45] respectively.

Centrifugal compressors are generally more efficient, smaller and lighter than their positive-displacement counterparts. Their drawback lies in the fact that the supplied boost increases with the square of the rotational speed, resulting in a low boost at low engine

[2] Philander and Frances Roots patented what is now referred to as the Roots blower in 1860 as a machine to force air into blast furnaces and mine workings. It thus predates the Otto cycle engine.

speeds [38]. This is ideal in aircraft and marine engines, which are commonly matched to a propeller [32], but not for an automotive engine which uses far more of its operating map. Furthermore, the pressurization requires high tip speeds and therefore in order to provide sufficient boost, centrifugal superchargers usually need to be driven by some form of step-up gearing which inevitably incurs some additional mechanical losses [46].

Until now, centrifugal superchargers have generally been aftermarket parts for passenger car engines, and one application, the BRM V16 racing engine, famously demonstrated the challenge of mitigating the speed-squared relationship between engine speed and boost pressure [47]. One means of alleviating the undesirable boost build-up characteristic is to employ a drive ratio-change mechanism. Such systems became commonplace on aero engines in order to better match performance to altitude [31-32, 48], and one engine, the Daimler-Benz DB601, used a continuously-variable speed-change mechanism, albeit with relatively high losses [31]. Recently two devices including SuperGen and V-Charge have been shown which achieve continuous variation of the drive ratio, and this capability has shown significant performance potential over a Roots-type device as the high-pressure stage in a compound charging system [46, 49]. **Figure 3** shows the Torotrak V-Charge device [46].

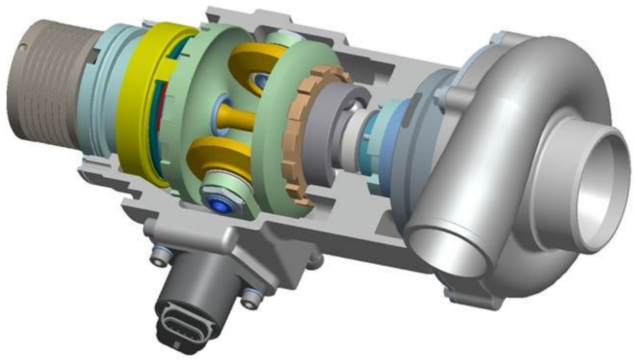

Figure 3. Torotrak V-Charge mechanism. [46]

Apart from external positive-displacement devices that provide boost for a passenger car engine, the engine cylinder itself can also be treated as a positive-displacement unit. The Scuderi engine shown in **Figure 4** is an example of such an approach which divides the four strokes of a conventional combustion cycle over two paired cylinders, one intake/compression cylinder and one power/exhaust cylinder, connected by a crossover port [50-51]. By dividing the function of compression and power into two paired cylinders in a 'split cycle', in theory the thermal efficiency will be improved due to the increased flexibility in optimizing combustion and pumping work. In addition, the possibility for pneumatic hybridization [52-53] and Miller cycle [54] can also provide additional fuel consumption benefits. However, pumping losses past the crossover valves and through the crossover ports, energy losses during the heat transfer from the compressed air to the compressed cylinder and crossover port walls and thermal loading are challenges for this kind of devices. Nevertheless, recent advances show promise, albeit with greater levels of complication [55].

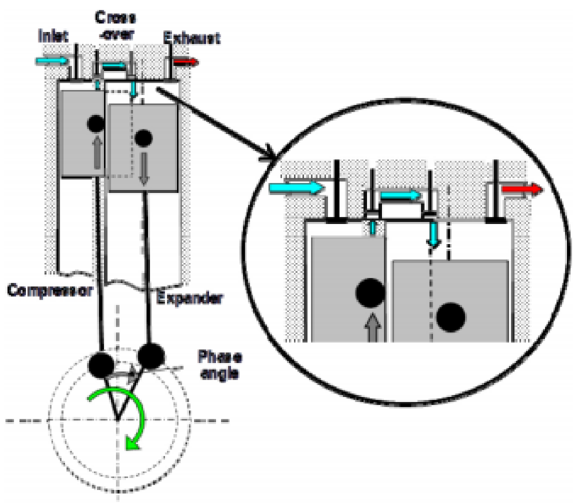

Figure 4. Principal elements of split cycle engine. [50]

The pressure wave supercharger (PWS) or sometimes termed as Comprex, seen in **Figure 5**, is an unconventional device that unlike any other supercharging system above pressurizes the intake air directly by the compression wave generated by the pressure difference between the exhaust and intake mass flow [56-60]. The transient response of a PWS-based gasoline engine is superior due to the fact that the pressure wave propagates at speed of sound. In addition, for some engine operating points the isentropic efficiency of the PWS system is much better than that of a turbocharged counterpart. For example, the maximum efficiency of a PWS was 75% at 3000 revolutions per minute (RPM) according to the research conducted by Oguri Y et al [60]. It might be noted that unlike any other conventional mechanical supercharging system whose power is directly provided by the engine crankshaft, the PWS system hardly consumes the useful engine power (more like a turbocharger), even though the rotor is usually driven by the engine. There are several challenges for the PWS system when it is adapted to a passenger car engine. For example, the length of the rotor channel cells, the port timing and the rotor speed all have to be matched to the engine size and can be adjusted for different engine operating points. In addition, the relatively lower isentropic efficiency of such a device at higher exhaust pressure is also need to be overcome before applying it to a production passenger car engine [60].

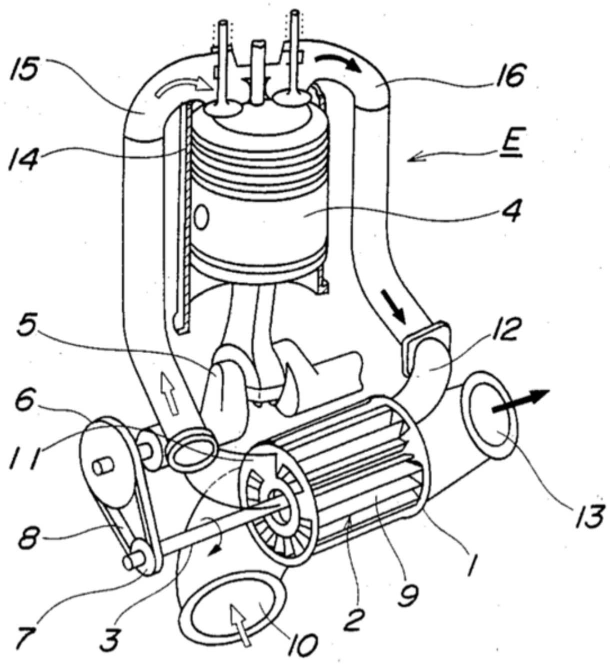

Figure 5. Pressure wave supercharger. [56]

1: rotor housing; 2: rotor; 3: rotary shaft; 4: piston of engine; 5: crankshaft; 6: crankshaft pulley; 7: rotary shaft pulley; 8: belt; 9: rotor cells; 10: air inlet; 11: air outlet; 12: exhaust gas inlet; 13: exhaust gas outlet; 14: cylinder of engine; 15: intake passenger; 16: exhaust passage; E: piston engine

3 Novel compressor designs

This section will discuss three compressor types that are new to the market: the TurboClaw, the Eaton Twin Vortices Series (TVS) V-Series, and a novel compressor from Honeywell. These compressors are able to enhance the low-end torque and improve the transient performance of a boosted downsized engine, thus providing the potential to further downsize and down-speed it. In the following, first the characteristics of Turboclaw are presented, followed by a critical discussion of the Eaton TVS V-Series and the novel Honeywell compressor.

The TurboClaw compressor is able to provide a higher boost at a lower speed range, by radically changing the geometry of the compressor to one with highly forward swept blades (see **Figure 6**) [61]. A Turboclaw compressor map can be seen in **Figure 7** [61]. It is similar to that of a conventional turbocharger compressor but it operates at significantly lower speed. Pullen et al. [62] demonstrated that by electrically supercharging a gasoline engine using TurboClaw compressor, which would not be possible for a conventional radial compressor, the low-end torque and the transient performance of the engine would be greatly improved.

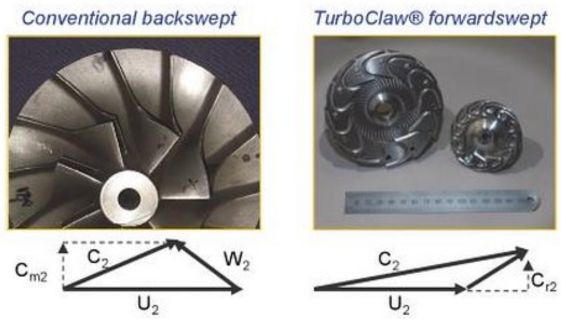

Figure 6. Turbocompressors and velocity triangles: Left: conventional backswept; right: TurboClaw forwardswept. [61]

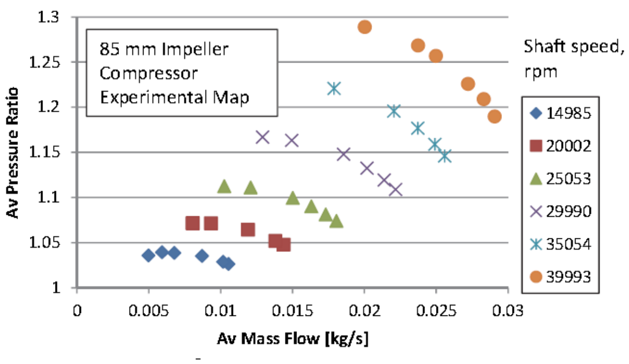

Figure 7. TurboClaw compressor map for an 85mm rotor. [61]

In conventional Roots-type positive displacement blowers, due to manufacturing tolerances, smaller clearances at multiple locations between two opposing supercharger rotors, between the rotor and housing (see **Figure 8**) and between the end plates and rotor ends (see **Figure 9**) cause leakage from the outlet of the conventional series-production Eton TVS R-Series to the transfer volume and from the transfer volume to the inlet, resulting in reduced volumetric efficiency. Specially, the smaller TVS R-Series devices, which typically match a 3- or 4- cylinder engine from 0.5 L to 2.0 L displacement, have substantially reduced low-speed volumetric efficiency due to the increased leakage area to displacement ratio. In order to improve the low speed volumetric efficiency, Eaton has optimized multiple parameters including rotor length to diameter ratio, the number of lobes, rotor twist and inlet/outlet port geometries. The output of several optimization iterations is a new TVS V-Series (an abbreviation of Volumetric Series). Compared to the R-Series, the V-Series features approximately 32% less leakage cross sectional area per unit displacement. In addition, V-Series devices have better isentropic efficiency at lower supercharger speed due to its physical characteristics. The combination of higher volumetric efficiency and higher isentropic efficiency in the low supercharger speed range allows the reduction of the minimum engine peak torque speed or meeting the low-end torque target with a lower drive ratio than that would be required by the R-Series, in turn reducing the intake mass flow recirculation at higher engine speed resulting in better pumping work. Note that the V-Series can also increase the low-end torque to further downsize a passenger car engine if the boost capability rather than engine knock was the constraint [44].

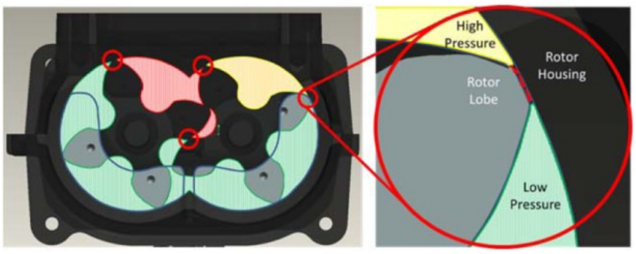

Figure 8. Diametral rotor leakage locations (left) and air leakage between the rotor lobe tip and rotor housing (right). [44]

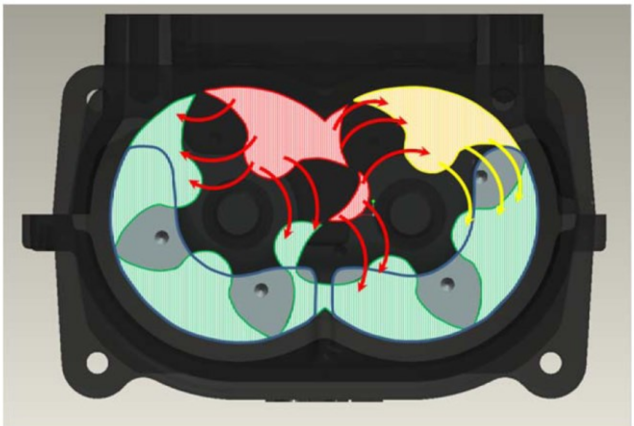

Figure 9. Leakage between rotors and end plates. [44]

The novel compressor supplied by Honeywell is also a device that can provide the same pressure ratio at lower rotational speeds [46]. It is not clear at the moment which technology has been implemented to achieve this from the literature. The deficiencies of this type of compressor are lower maximum pressure ratio and narrower speed range. The merits of such a device when fitted on a passenger car engine are similar with that of the TVS V-Series, i.e. reducing the minimum engine peak torque speed if the same drive ratio is applied or reducing the drive ratio if the same peak torque speed is desired in turn minimizing the pumping losses at higher engine speed. Both scenarios can help to further downsize or down-speed a gasoline engine [46].

4 The relationship between downsizing, driveability and down-speeding

Engine downsizing refers to the use of a smaller displacement engine to replace a larger displacement engine, usually employing turbocharging or supercharging in order to maintain in-vehicle installed power [8-9]. Most downsized passenger car engines currently offered in the market place appear to have a 'downsizing factor' of approximately 35% to 40% [10]. The downsizing factor (DF) is defined to as

$$DF = \frac{V_{Swept_{N/A}} - V_{Swept_{Downsized}}}{V_{Swept_{N/A}}} \qquad (1)$$

Where V_{Swept} is the swept volume of the engine, and the subscripts N/A and Downsized refer to the naturally aspirated and downsized engines respectively.

It is anticipated that the DF of future highly downsized engines may reach levels at or above 60% as some current prototype engines can now achieve approximately a DF of 50%-60% using some new technologies currently in development. This is mainly due to the demand to reduce vehicle CO_2 emissions further in the near future. Practically, this can only be achieved by a two-stage boosting system along with at least one turbocharger to make use of waste exhaust gas energy [63]. Such two-stage configurations can also realize higher overall isentropic efficiency as both the high- and low-pressure boosting devices are able to be operating in their higher isentropic efficiency area in two different flow regimes.

The motivation for further downsizing a passenger car engine can also be seen in **Figure 10**, which shows the potential for downsizing based on some validated data [3,10]. The increase in improvement is mainly due to the reduced pumping losses at part load, reduced heat losses and lower friction resulting in higher mechanical efficiency for smaller displacement engines.

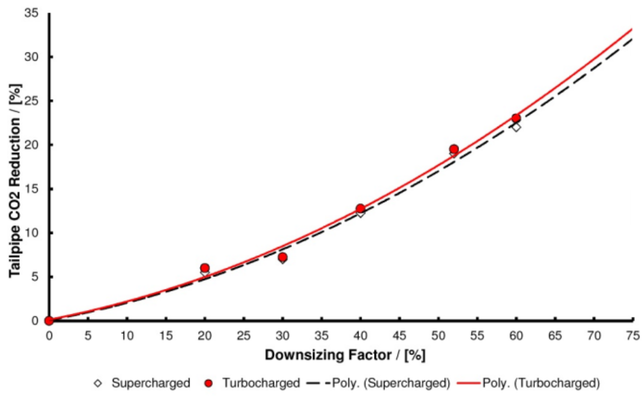

Figure 10. The potential for downsizing based on validation data from earlier work. [3]

However, appropriate engine downsizing is also important when a driving cycle fuel consumption and driveability are considered. The so-called 'right sizing' of the engine should be able to handle the power requirement of the frequent operating points without the assistance of boosting devices. This is especially true for a fixed-drive-ratio supercharger system equipped with a clutch, due to the fact that a large amount of parasitic loss would occur when only little boost is required and the driveability quality would be greatly impaired when a clutch is transiently engaged [64].

Driveability, defined as the degree of smoothness and steadiness of acceleration of an automotive vehicle in this paper, describes the driver's expectation of a vehicle. It is, by its nature, a subjective rating, and hence is difficult to quantify. However, there seems to be some correlation between subjective assessments and objective measurement (characteristic values can be seen in **Figure 11**) of a vehicle's behaviour. To be more specific [65]:

a) There is a strong correlation between delay time/initial acceleration and launch feel, while there is no clear correlation between jerk and launch feel.

b) There is a clear correlation between delay time/initial acceleration/jerk and performance feel.

c) There is a degree of coupling between delay and initial acceleration in the subjective assessment of launch and performance feel.

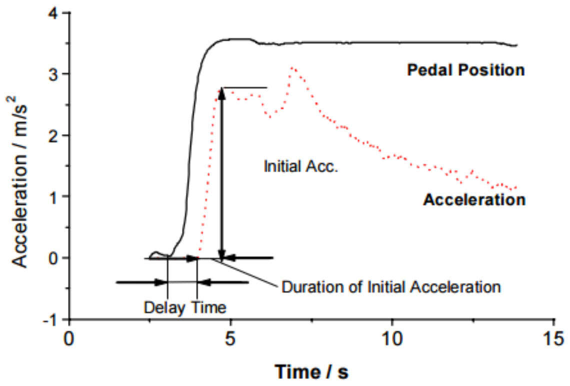

Figure 11. Illustration of the characteristic values. [65]

Delay time: the time between the first change in pedal position and the first change in the acceleration trace;

Acceleration: peak value of the initial acceleration phase;

Jerk: the value of the initial acceleration divided by the duration of the initial acceleration.

Boosted and especially turbocharged engines are considered to have poorer driveability than their naturally-aspirated counterparts due to the perceptible time needed for the boosting system to generate sufficient intake mass flow. With the requirement of further downsizing, the driveability issues are growing more severe. In this context, supercharging might have to be introduced either in a single-stage configuration or in a compound charging arrangement in order to overcome turbo-lag at low engine speeds.

Since downsizing is arguably already reaching a limit [10], down-speeding may become much more important for increasing fuel efficiency in the near future [66]. Down-speeding refers to lowering engine speeds by means of using longer gear ratios or via the optimization of the transmission gear shift strategy to further improve the vehicle fuel economy of a downsized engine. In many respects, down-speeding functions similarly to downsizing, i.e. it moves the engine operating points to a higher efficiency region on the engine characteristic map. However, down-speeding can be expected to negatively affect a vehicle's transient performance, thus for a particular application the driveability needs to be taken into consideration before optimizing the transmission gear ratio or the strategy.

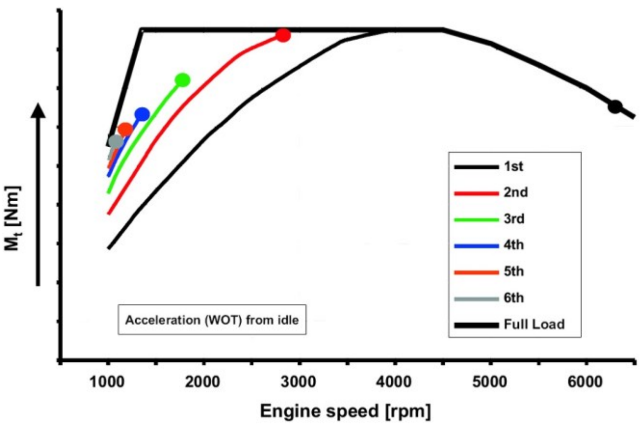

Figure 12. Comparative analysis of transient vs steady state engine torque delivery for a turbocharged engine: M_t achieved after 5 seconds in different gears vs steady state. [67]

Note that it might not be sufficiently fair to compare the fuel economy of different boosting systems using only the same engine operating points, as in a real driving cycle the different transient characteristics of different boosting configurations coupled with different transmission shift strategies will bring the engine to different operating points. For instance, **Figure 12** shows the engine torque and speed trajectory after five seconds of full load acceleration from near idle speed in each gear for a turbocharged engine [67]. Thus, a relative accurate fuel economy calculation should be based on the engine transient performance and transmission events. In addition, fuel economy comparisons would be better conducted for similar vehicle performance metrics, such as a similar time interval from 0 to 100km/h. In the following, first the effects of down-speeding are studied by means of using longer gear ratios, followed by some down-speeding control that is achieved by the optimization of the transmission gear shift strategy. In both situations, the vehicle was operated in a dynamic environment and the vehicle performance metrics were maintained.

Birckett et al. [68] simulated and tested a mechanically-supercharged 2.4 litre in-line 4-cylinder gasoline direct injection engine employing the Miller cycle and a high compression ratio. It was proved that down-speeding, enabled by altering the transmission ratios (compared to downsizing the engine, in combination with supercharger de-clutching, compression ratio augmentation and Miller cycle, and cooled exhaust gas recirculation (EGR)), contributed most to the fuel economy improvement over the 2.4 Litre naturally aspirated baseline. However, it is worth noting that the low speed torque capacity was increased by approximately 100Nm and that without the enhancement of the low speed torque and the corresponding increase in transient capabilities, the vehicle performance would have been greatly affected [66].

Wetzel [66] compared three different two-stage boosting systems including twin-sequential-turbocharger, supercharger-turbocharger and turbocharger-supercharger configurations while maintaining the transmission and final gear ratio in simulation. It was shown that at an engine level, the twin-sequential-turbocharged configuration had slightly lower full-load brake specific fuel consumption (BSFC) whilst at the vehicle level the supercharging boosting systems benefitted from approximately 8%-10% lower BSFC over the New European Driving Cycle (NEDC) and 12%-14% over the Assessment and Reliability of Transport Emission Models and Inventory Systems (ARTEMIS) urban driving cycle when compared to the twin-sequential-turbocharged configuration, due to its capability to enable down-speeding.

Ostrowski et al. [69] studied the effects of down-speeding and supercharging a passenger car diesel engine in test and simulation. Their results suggested that transmission shift schedule optimization can show greater fuel consumption benefits than the down-speeding via changing the drive ratios. In addition, after the transmission optimization, their in-vehicle simulation results of the supercharged configuration showed up to 12% fuel economy improvement over that of the turbocharged counterpart, along with a corresponding reduction in transmission shift frequency of up to 55% while maintaining the same first gear acceleration, top gear passing and 0-60 mph acceleration performance.

Based on the analysis above, it is clear that there is a strong relationship between downsizing, driveability and down-speeding. For boosted engines specifically, further downsizing can be enabled by the assistance of supercharging in a compound charging system; The driveability issue of a downsized engine can be mitigated by proper matching of a supercharger; and from the perspective of the real driving fuel economy, downsized passenger car engines could employ supercharging technology to further move the engine's operating points into a more efficient area whilst maintaining similar vehicle driveability.

5 Mass production and prototype downsized supercharged passenger car engines – an overview

There is a number of mass production and prototype downsized supercharged passenger car engines, listed in **Table 1**, to satisfy various customer requirements currently. They all have enhanced specific power and low-end torque along with improved transient performance compared to a similar-size turbocharged counterpart. Note that among the list, the supercharger type is almost positive displacement, and most are Roots-type; and an active valve is usually employed to bypass or 'recirculate' the flowing through the supercharger when boost is not required. In addition, approximately half of the listed supercharged systems have a clutch equipped and the clutched configurations have a larger drive ratio than that of the systems without a clutch.

Table 1. Typical downsized and supercharged passenger car engines

Ultraboost (target) [10]	Ricardo HyBoost [70]	Ford 1.0L ECOBOOST – V-Charge [46]	Volkswagen 1.4TSI [71]	Volvo T6 [72-73]	Jaguar Land Rover AJ126 [74]	Audi V6 TFSI [42]	Mercedes-Benz C32 AMG [75]	Mazda KJ-ZEM [76-77]	Nissan HR12DDT [78]
142 (6500 RPM)	105 (5500 RPM)	-	90	113	92	71	81	70.5	60
32 (3500 RPM)	29 (2500 RPM)	-	22 (1750-4500 RPM)	25 (2000-45000 RPM)	19 (5000 RPM)	17.5 (2500-4850 RPM)	17.7 (3000-4600 RPM)	16 (3500 RPM)	15
2.0	1.0	1.0	1.4	2.0	3.0	3.0	3.2	2.3	1.2
9.0	9.0	9.0			10.5	10.5	-	10.0	13.0
Two stage supercharge r + FGT	Two stage electric supercharger + FGT	Two stage supercharger + FGT	Two stage supercharger + FGT	Two stage Supercharger + FGT	Single stage supercharger	Single stage supercharger	Single stage supercharger	Single stage supercharger	Single stage supercharger
25	23 (with electric supercharger assisted)	-	16	15.7	12	-	-	-	-
Better than JLR 3.0L Twin Turbo V6 Diesel	-	At 1100 RPM, from 2 bar BMEP to full load, engine torque rising to 90% of the target full load within 0.73s.	At 1250 RPM, from 2 bar BMEP to WOT, 2 bar intake manifold pressure achieved within 2.5s	-	At 1000 RPM, from 1 bar BMEP to full load, engine torque rising to 90% pedal position within 2.3s	-	-	-	-
5.9	-	Step-up: 3 CVT:0.281-2.82 Epicylic: 12.67			2.483	2.5	-	-	2.4
Yes	-	No	Yes	Yes	No	No	-	No	Yes
Prototype	Prototype	Simulation-phase	Production	Production	Production	production	Production	Production	Production
Eaton R-series R410	Centrifugal	Honeywell Compressor	Eaton	Eaton	Eaton R-series R1320	Eaton R-series R1320	Teflon-coated rotors Twin-screw	Lysholm compressor	Eaton Roots
Active		Passive	Active	Active	Active	Active	Active	Active	Active

	Specific power (kW/l)	Peak BMEP (bar)	Displacement (l)	Compression ratio	Boosting configuration	BMEP at 1000 RPM (bar)	Transient response	Drive ratio	Clutch	Engine Type	Supercharger Type	Bypass Type

TSI: twin-charged stratified injection; BMEP: brake mean effective pressure; JLR: Jaguar Land Rover; BSFC: brake specific fuel consumption; CVT: continuously variable transmission; FGT: fixed geometry turbine; TVS: Twin Vortices Series; TFSI: turbocharged fuel stratified injection; RPM: revolutions per minute.

6 Challenges of conventional supercharged passenger car engines

Clutch control:

In a compound charging system, at high engine speeds, for a supercharger with a fixed ratio drive configuration, both positive-displacement and centrifugal devices might violate their maximum speed limits due to their corresponding physical characteristics. In addition, for the low load within the naturally-aspirated region, constantly connecting a supercharger to the crankshaft will increase the engine's parasitic losses which will inevitably result in a degraded fuel efficiency. In this context, a clutch may be a necessary component for a supercharged engine in order to be disengaged at high engine speed and steady-state low-load operating regions and to be engaged at low-speed high-load operating points and when a transient signal is triggered [46].

Note that in **Table 1** that Jaguar Land Rover AJ126 3.0L V6 does not use a clutch as standard and according to Meghani et al. [74] de-clutching only delivered a peak improvement of 2.6%. This might be due to the fact that the clutch was mounted on the supercharger nose, i.e. after the drive pulley and belt system, and thus a large amount of associated parasitic losses still existed. In the non-clutched configuration, the control strategy will be much easier to implement as only a throttle or a combination of a throttle and an active bypass valve is needed to be controlled. This will also have some other benefits such as improved noise, vibration and harshness (NVH) and more compact packaging. However, the effect of the increased temperature after the supercharger due to the recirculation of the intake mass flow needs to be taken into consideration, especially when a turbocharger is installed after the supercharger to provide enhanced boost capability and possibly better fuel efficiency. Another factor worthy of note is that the non-clutch configuration might only be feasible for the positive displacement configurations, since for a fixed-drive-ratio centrifugal counterpart, due to its characteristics i.e. that its boost increases with the square of the rotational speed, a possibly larger parasitic losses will be incurred.

A fully-developed boost control (with or without clutch control) is a key technology to implement a supercharger into a passenger car engine or a vehicle as part of a compound charging system when the supercharger has a high drive ratio in the interests of generating high boost at low engine speed. There seems no literature covering this area to the authors' knowledge. However, based on some simulation and test projects conducted at the University of Bath [10, 38, 46, 49, 65, 79, 80], it is still possible to summarize some key challenges for developing a viable control strategy for the supercharger system (note that these follows only some suggestions arising from the engine simulation and test results from the University and these should not be interpreted to represent or be calibrated for any specific application).

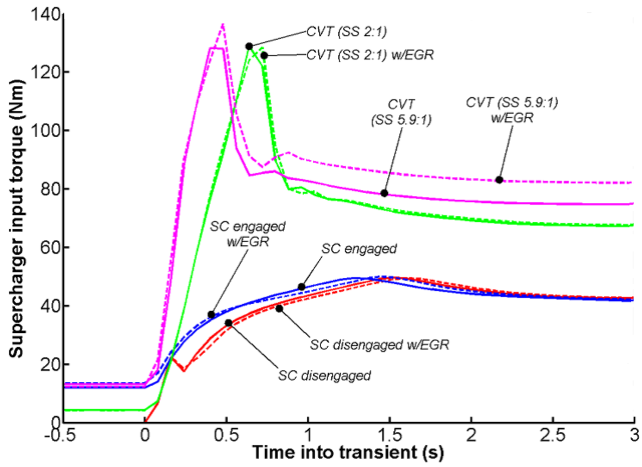

Figure 13. Supercharger input torques for tip-in simulations in the supercharger-engaged regime, the supercharger disengaged regime and for a CVT-driven supercharger regime. [80]

CVT: continuously variable transmission; SS: steady-state ratio; w/EGR: without exhaust gas recirculation; SC: supercharger.

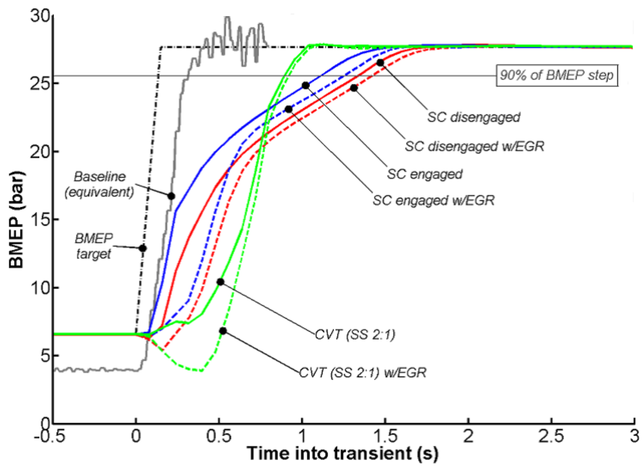

Figure 14. BMEP response for tip-in simulations of the supercharger engaged regime and the CVT-driven supercharger regime, showing the effect of the initial

steady-state CVT ratio. [80]

For reference, the BMEP target, 90% of the BMEP step demand (i.e. full load), and the equivalent BMEP for the baseline experimental results are also shown.

BMEP: brake mean effective pressure; CVT: continuously variable transmission; SS: steady-state ratio; w/EGR: without exhaust gas recirculation; SC: supercharger.

First of all, if a clutch is mounted, having the clutch always closed (termed Sport Mode below) provides the best scenario for the transient performance due to the elimination of the transient delay of this element, but the fuel efficiency is also largely compromised. Economy Mode refers to when the clutch is disengaged during the steady-state low load operation and clutched in when boost is demanded. Economy Mode requires selection of a proper clutch engagement time, as a shorter time will either break the clutch due to the unavoidable torque (see Equation 2) or generate a torque dip (due to the referred inertia of the supercharger rotors) during a transient event affecting driveability. For example, in the simulation study by Rose et al. [80], an amount of clutch acceleration torque can be seen in **Figure 13**. The engine torque dip phenomenon can be seen in **Figure 14**, in which it can be seen that the economy mode (marked SC disengaged) has significantly less engine torque during the first part of a transient event compared to the sport mode (marked SC engaged in the figure). A longer engagement time is also undesirable since this negatively influences the engine's time-to-torque performance also resulting in a degraded driveability. For the active-bypass-valve configuration, the timing to close the valve is also important during a transient; too fast a response will impede the gas transfer when the supercharger is still rotating slowly after the closing of the bypass valve. Thus a considerable time is needed to calibrate the system before a good transient response and acceptable driveability is achieved.

$$T = I \times \dot{\omega} \qquad (2)$$

T: torque; I: inertia; $\dot{\omega}$: angular acceleration

Secondly, given the context that some automotive manufacturers partially close the wastegate of their turbocharger at part load to trade some fuel-efficiency for an enhanced transient performance, the supercharger could also be boosting within what is normally the naturally aspirated region or, if a compound charging system is considered, the turbocharger is boosted at the steady-state low-load operating point to take the trade-off between part-load fuel efficiency and transient performance into consideration. For the positive displacement supercharger fitted with a clutch, the supercharger pre-spin strategy can also be utilized. It is when the bypass is closed with the clutch still disengaged, engine pumping speeds up the rotors to approximately 1/3 of the engagement speed, minimizing engagement time [81].

Finally, the ability to hand over the boost generation from a supercharger to a turbocharger is also challenging for a twin-charged passenger car engine. Also note that at high engine speeds when the fixed-ratio supercharger is not operational, a longer transient response is anticipated which will deliver an inconsistent driveability characteristic [49].

<u>Bypass valve type selection:</u>

For a positive-displacement, fixed-drive-ratio supercharger system, an active supercharger bypass might be a necessary component to fulfil the function of bypassing the supercharger when it is not needed if a clutch is fitted and controlling the boost. This is illustrated in **Figure 15**; although throttle control is able to alter the engine torque this will incur generating a larger amount of driving torque for the supercharger. In addition, an active bypass system allows linear air control, seen in **Figure 16**, which is considered to be beneficial for noise suppression [78]. However, as discussed, adopting an active bypass valve will bring extra complexity to the control system and might require a considerable time of calibration before the supercharger system can function for the whole engine possible operating regions.

Although some passive bypass valves can function similarly like active bypass valves by tuning the action of the valve (Lotus using a passive, boost-capsule-operated bypass, for example [41]) i.e. it opens on throttle lifts and idle, progressively closes as the throttle is opened and opens partially at peak boost to control the level reached, they might suffer some fuel consumption penalty at part load. In addition, the passive bypass valve may not be able to effectively use the supercharger pre-spin strategy, the operation of which is to close the active bypass valve with the clutch still disengaged to allow the engine pumping the supercharger rotor speeds to approximately 1/3 of the engagement speed, minimizing engagement time [81].

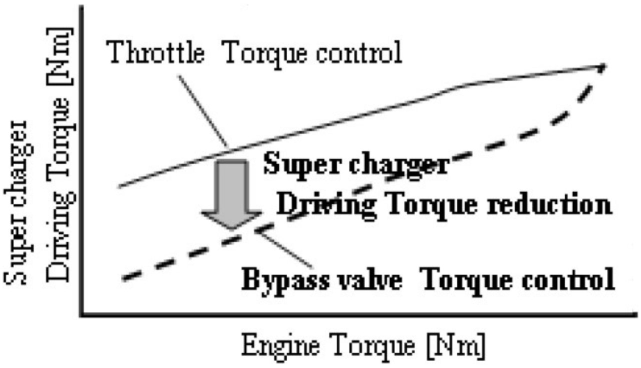

Figure 15. Effect of bypass valve control. [78]

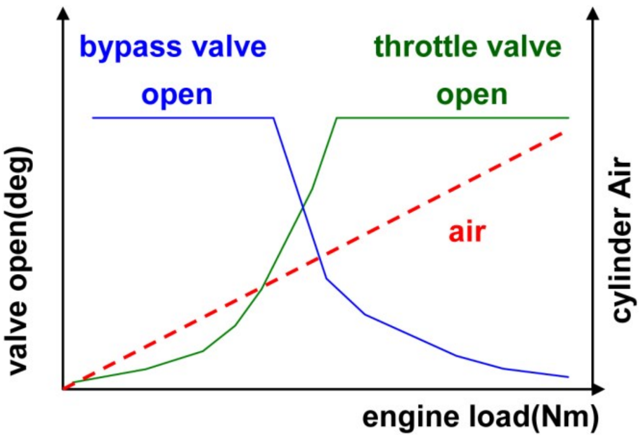

Figure 16. Throttle valve and bypass valve motion. [78]

Regarding boost control specifically, a large amount of intake mass flow will be recirculating around the supercharger via the bypass valve at high engine speed and part load, resulting in reduced fuel efficiency; this also affects the intake manifold temperature which might trigger knock. Although the reduction of drive ratio will mitigate the recirculation phenomenon at high engine speeds, low-end torque might suffer.

7 Potential trends for mechanically supercharging a passenger car engine

In order to overcome the deficiencies in the performance and driveability of production supercharged engines, its development has been continuous. Low-end torque enhancement, transient driveability improvement and low-load parasitic loss reduction are three of the major development directions, and there is a degree of coupling between them. For example, an improved low-end torque will also help the time-to-torque transient performance.

<u>Low-end torque enhancement:</u>

As mentioned earlier, selecting some ‘low speed compressor’ will increase the low-end torque, due to its capacity to provide a larger pressure ratio (PR) at low engine speed. For instance, Meghani et al. [74] replaced the Eaton R-Series R1320 with a new Eaton V-Series V1270C in their Jaguar Land Rover AJ126 3.0L V6 engine. Their test results indicated that even when equipped with a reduced drive ratio (from 2.483 to 2.472) and with a lower swept volume the device can deliver a noticeable torque gain at low engine speed. This is mainly due to the increased volumetric efficiency of the Eaton V-Series device and possibly also enhanced isentropic efficiency. However, they also pointed out that with the clutch engaged, due to the more severe recirculation of the intake mass flow, the fuel consumption would increase rather than decrease at low load, where an improvement would be expected to come from the reduction of the drive ratio. Hu et al. [46] studied a similar configuration using a novel compressor supplied by Honeywell. This compressor was also claimed to have a higher boost capability at low engine speeds and its peak isentropic efficiency occurred in the low speed and pressure ratio area. Their simulation results indicated that by using the same drive ratio, this novel compressor configuration can increase the low-end torque performance. In addition, if the same BMEP is maintained, the novel compressor still benefitted from an enhanced isentropic efficiency which is important in designing a supercharger-turbocharger system. They also showed that in the Torotrak V-Charge variable drive supercharger system, this compressor had better transient response compared to a conventional centrifugal compressor when the other control parameters were fixed. As such, their findings were in line with those of Turner et al., who investigated the SuperGen electromechanical supercharger in a highly-downsized compound-charged engine application [49].

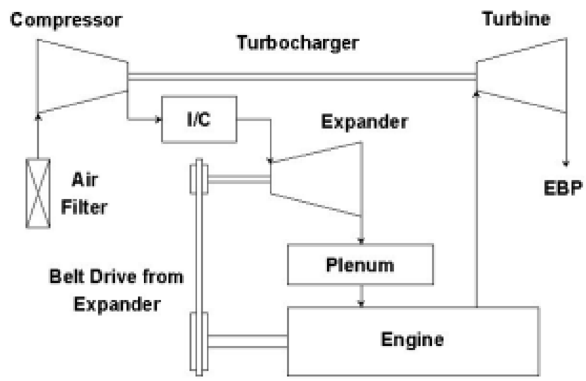

Figure 17. Schematic of Turbo-expansion concept. [84]

EBP: exhaust back-pressure

Increasing the drive ratio is another option to increase the low-end torque. However, a fixed-drive-ratio unit might not be suitable in this case, as a higher drive ratio will cause more intake mass flow recirculation at high engine speeds resulting in higher parasitic losses [82]. The high drive ratio might also bring the supercharger to the over-speed region at a lower engine speed, necessitating it being clutched out earlier in compound charging systems in turn resulting in less boost assistance at higher engine speed. Thus, a continuously variable transmission (CVT) might be preferable to adjust the drive ratio for different engine operating points.

Several CVT-driven supercharger applications have been studied in the literature. They have all been demonstrated to markedly enhance a supercharged engine's low-end torque. However, there are also some other trade-offs to be considered. For example, the research by Rose et al. [80] focused on the relationship between part-load efficiency and transient response in a highly boosted downsized gasoline engine. Hu et al. [83] and Turner et al. [84-85] on the other hand studied the Turbo-expansion concept (as can be seen in **Figure** 17) by treating the positive-displacement device as an expander. It was shown that if a positive-displacement supercharger's speed could be reduced far enough (CVT preferred), the supercharger will move to an 'expansion mode' from its conventional compression mode, as seen in **Figure 18**. However, the isentropic efficiency of a compressor in its 'expansion-mode' will drop significantly to approximately 45% as seen in **Figure 19** [86]. In simulation (although the later test results were not realistic [85]) Turner et al. [84] claimed that that by placing the supercharger at the high-stage of a compound charging system and over-compressing the turbocharger system by further closing the wastegate whilst using the supercharger as an expander to reduce the boost pressure after the supercharger to that required, there would be some BSFC benefits as some of the otherwise wasted exhaust energy can be reclaimed directly from the supercharger to the engine. The combustion efficiency could also be increased as the intake manifold temperature was reduced by the 'expansion effect' which might enable advancement of the spark timing depending on the Residual Gas Fraction (RGF) increase resulting from the higher exhaust back pressure. The undesirable test results finally achieved by Turner et al. [85] were studied by

Taitt et al. [87]. They proved that for the components used on the investigated engine, it was the low isentropic efficiencies that prevented the necessary temperature reduction of the inlet charge air.

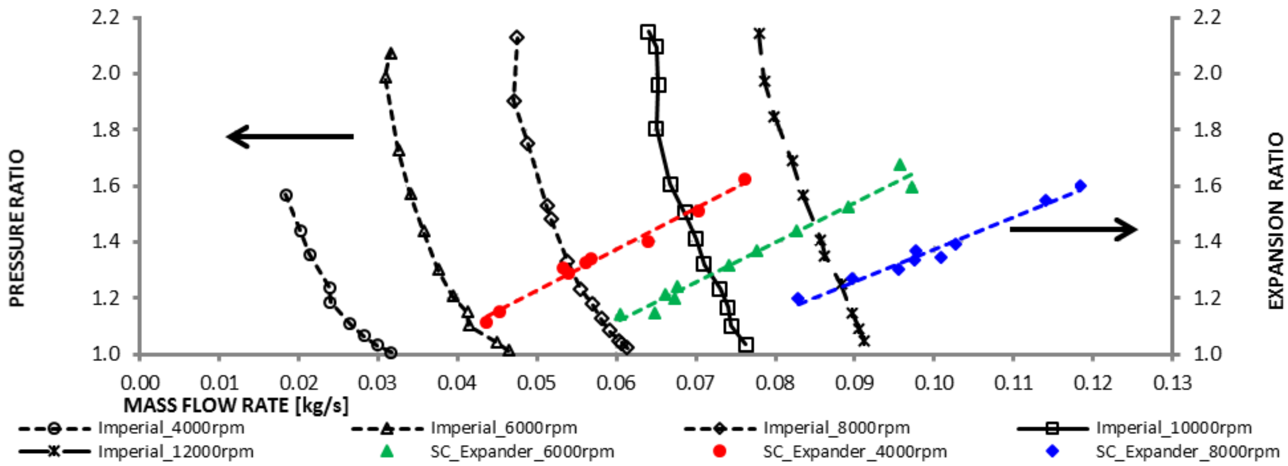

Figure 18. Pressure and expansion ratio of the Eaton R-Series R410 supercharger versus mass flow. [86]

RPM: revolutions per minute; SC: supercharger.

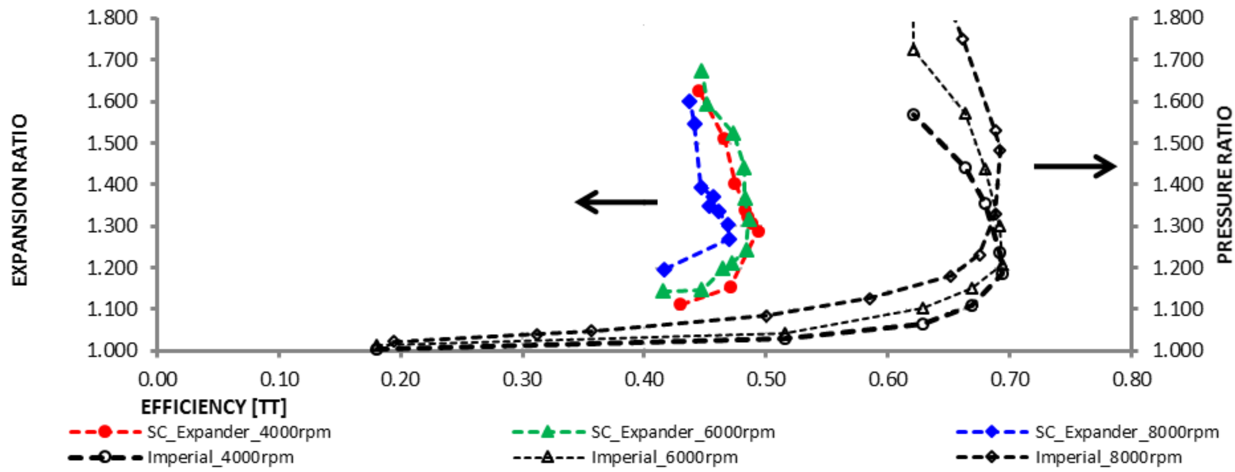

Figure 19. Total to total isentropic efficiency of the supercharger operating as a compressor and expander. [86]

RPM: revolutions per minute; SC: supercharger.

Transient driveability improvement:

Firstly since a transient clutch engagement event will bring some NVH issues, a non-clutch configuration seems to be an ideal option. However this will result in reduced fuel efficiency at part load when boost is not needed, due to increased pumping work through the bypass valve (or the throttle if it is the only control parameter) and larger mechanical losses via constantly connecting a supercharger with the engine. More importantly, supercharging a passenger car engine without a clutch might over-speed the supercharger or affect its reliability and durability. In this context, a CVT-driven supercharger shows its advantage, as by altering the CVT ratio, the supercharger rotor speed can be kept within its prescribed limits, also resulting in less parasitic losses at part load. The research by Hu et al. [46] has proved this. Their simulation results suggested that a clutch might not necessarily be needed for a CVT-driven centrifugal compressor, as at low load by reducing the CVT ratio there was only approximately 2% fuel consumption penalty and at higher engine speeds the supercharger was well within the speed limit along with only approximately 0.5% fuel consumption consumed to drive the supercharger constantly.

Note that the CVT-driven supercharger (both positive-displacement and centrifugal) can also help the boost handover (from supercharger to turbocharger) during the end of a transient for a twin-charged passenger car engine and mitigate the inconsistent driveability characteristic at high engine speed when the fixed-ratio supercharger is not operational [49].

Secondly, a compressor suitable for generating increased boost at low engine speeds can also help improve the transient driveability due to its quicker boost generation ability. A CVT inherently provides this capability, given that its ratio range is wide enough. For the detailed analysis of this effect on an engine, it is suggested to refer to the previous section.

Finally in this section, some novel mechanisms to provide supercharging are discussed. The followings are organised by the category of purely mechanical supercharging to purely electric supercharging. As this work is focused on mechanical supercharging, electric supercharging is only briefly introduced to discuss the advantages and deficiencies of a mechanical supercharger system compared to an electric counterpart. Note that as there is a degree of coupling between the torque capability and transient response, the mechanisms discussed below are all beneficial for the engine’s low-end torque.

Lontra's Blade Supercharger is a variable flow compressor with the ability to meet the boosting requirements of a heavily downsized engine. **Figure 20** shows this positive displacement rotary device with a simple variable inlet port that provides dynamic control of the air mass flow rate and internal compression ratio without changing its rotational speed [88].

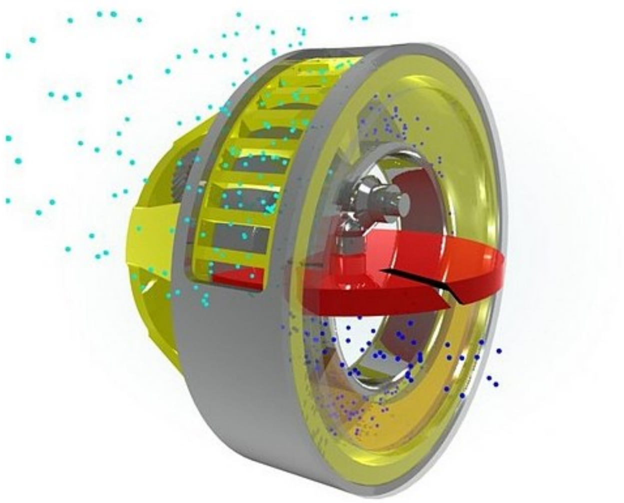

Figure 20. Lontra blade supercharger. [88]

The VanDyne SuperTurbo is also an enabling technology for heavily downsizing an engine without loss of vehicle transient response and peak power [9]. This technology, with its schematic shown in **Figure 21**, can achieve the benefits of turbocharging (via 1-2-3-4 in the figure) and supercharging (via 6-5-3-4) and turbocompounding (via 1-2-3-5-6) by uniquely controlling the CVT ratio.

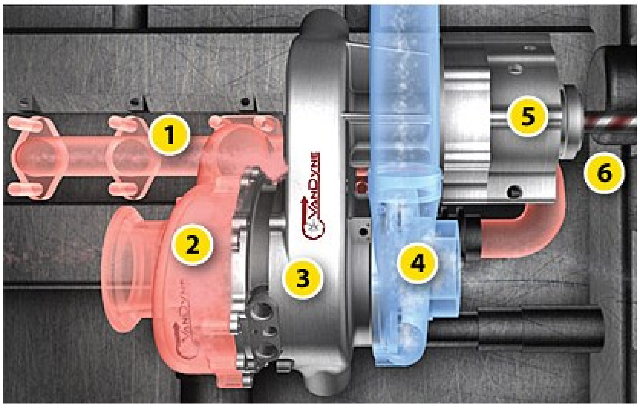

Figure 21. VanDyne SuperTurbo [89]

1: exhaust manifold; 2: uniquely designed turbine; 3: high speed drive; 4: matched compressor; 5: continuously variable transmission; 6: engine interface

Turner et al. [49] described the performance of the SuperGen device on an extremely downsized gasoline engine (Ultraboost in **Table 1**, DF = 60%) compared to a fixed-ratio positive-displacement counterpart (the Eaton TVS R410 used in original development of this engine [10]). The basis of the SuperGen supercharger is a power-split electromechanical transmission technology with an epicyclic traction-drive and two small permanent-magnet motors that provide a fully variable transmission between the engine front-end accessory drive (FEAD) belt and the high-speed radial flow supercharger impeller (see **Figure 22**). Note that in the work of Turner et al. the system voltage was 12 V; this is possible since the energy path into the device is mostly mechanical (some power can be taken from a battery for transients, but this is only a small proportion of the whole) [49]. It was summarized in their work that at a steady-state 1000 RPM, SuperGen could enable a significantly higher torque by approximately 26.5% than its fixed-ratio positive displacement counterpart. The transient performance was also proven to be superior with an improvement of approximately 68%. Lastly since it is, in essence, a variable-ratio-drive device, a better match for the compressor speed was achievable which could provide 1.3% to 4.3% fuel efficiency improvement at part load conditions, due to the elimination of recirculation and wind age losses associated with the roots blower. Some of this improvement was also attributed to the use of a centrifugal compressor, which the drive mechanism permitted.

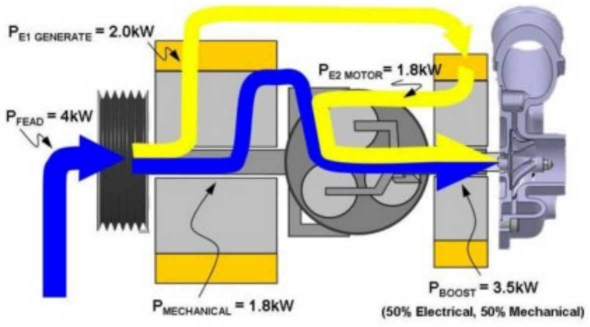

Figure 22. Power-split electromechanical transmission system of the SuperGen supercharger. Approximate power flow is shown for a mid-load condition. [49]

E-boosting, which is realized by electrically driving a compressor (usually centrifugal), is able to decouple the boost process from the engine operating point. It is usually used in series with a turbocharger to increase low-end torque and reduce turbo-lag. Typically an e-booster runs to approximately 1.4 bar-1.6 bar absolute on a standard 12 V setup within 200 ms – 500 ms [90]. For example, the Ricardo's HyBoost project (the schematic can be seen in **Figure 23**) showed that by using a 12 V unit, the electric supercharger could provide boost generation twice as fast as without an electric supercharger [70]. This is mainly due to the fact that the higher engine exhaust mass flow, achieved by the electric supercharger assistance, releases more power to the exhaust thus accelerating the turbocharger's run-up.

It is worth noting that some, if not all, of the energy used to drive the supercharger is essentially free (assuming that it is gathered from regenerative braking via the alternator and stored in the battery) and such a device is able to completely eliminate the torque dip issue challenging the control strategy. There are also other benefits from employing such a micro hybrid-type configuration (a total 12.5% fuel efficiency improvement was simulated during the HyBoost programme), including smart charging, improved stop-start and enhanced torque assistance. However, a potential drawback is that it is challenging to produce an e-booster that can run continuously due to design considerations regarding overheating the motor (if it is high powered) and the subject of battery depletion in-vehicle during extended periods of operation. If it can run continuously and the alternator can supply sufficient power, the round-trip power transmission losses can then be significant. In this area the SuperGen approach shows merits, since during steady state operation all power to the compressor comes into the machine from the FEAD, and the mechanical energy path through the device (which can transmit a significant proportion of the power – see **Figure 22** [49]) should be more efficient, while the electrical path does not go through the battery.

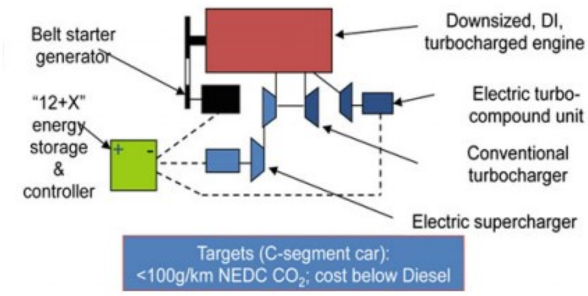

Figure 23. HyBoost concept scheme. [70]

As voltages are set to increase to 48 V so the efficacy of such an integrated approach improves on two levels: greater amounts of regenerative energy can be gathered via the higher-powered alternator, and the e-booster can generate more air flow and boost for longer (due to the greater amount of energy gathered, and assuming the battery has the capacity to support this). At this point the technology becomes viable in larger and heavier vehicles.

However, employing an e-booster at higher system voltages does assume that original equipment manufacturers (OEMs) will be able to manage the transition from 12 V to 48 V across multiple engine and vehicle platforms simultaneously, however; in 'old' 12 V vehicle platforms attempting to deploy a 48 V engine platform will not be without its difficulties – extra work on the electrical system, employing a 48 V sub-net in the engine bay for example, may be necessary. Again, as an alternative to this a SuperGen-type device, which is voltage agnostic (because the drive energy is supplied via the FEAD with only limited and occasional draw from the battery), would appear to be an ideal bridging technology to e-boosting as well as a complete solution to the problem, capable of delivering higher boost pressures because of its mechanical drive path.

As a final point when considering e-booster systems as part of a mild hybrid configuration with increased voltage, Aymanns et al. [91] investigated the merit of using energy stored in a battery to drive a supercharger versus applying it directly to the wheels via an electric motor. They reported that, because of the addition of fuel energy to the air supplied by the e-booster and the subsequent combustion providing increased work to the crankshaft, e-supercharging could provide approximately 10 times the torque to the wheels that just using the electrical energy in a motor could. Depending on driving situation, there is thus a trade-off to be investigated between the two means of utilization of regenerated energy in a drive cycle or real-world operation of a vehicle.

<u>Low-load parasitic loss reduction:</u>

As mentioned above, a supercharger system without a clutch will suffer some fuel economy reduction in the low-load region. It seems that the parasitic losses will always exist even though the drive ratio and the transmission efficiency could be further improved. However, Hu et al. [92] firstly proposed a novel concept to completely eliminate the parasitic losses at part load without the use of a clutch. They extrapolated the test data seen in **Figure 18** and **Figure 19** and imported the data into their twin-charged gasoline model. By controlling a CVT-driven unit, it is demonstrated that the conventional throttle control could be replaced by the supercharger CVT ratio control (the supercharger is now in its 'expansion mode') while still reclaiming some of the throttling losses that is otherwise wasted during the throttle control. Since no clutch is needed, the NVH issues of supercharging a gasoline engine will also be greatly improved. Other benefits such as the simplified control and the reduction of the intake manifold temperature could also show their potential. Nevertheless, some major deficiencies will have to be overcome before applying this concept to a production engine. They include the requirement of a large CVT ratio range (as wide as 30) and the consideration of the condensation effect, which might have some effect on the combustion behaviour. Meghani et al. [74] tested this concept by adopting a fixed-overall-ratio of 0.74. Four engine operating points including 1000 RPM 2.5 bar BMEP, 1500 RPM 8.3 bar BMEP, 1750 RPM 8.3 bar BMEP and 2000 RPM 5 bar BMEP were compared. The result trends are consistent with the simulation study by Hu et al, although they used a different Eaton supercharger on their test engine (Eaton V1270C V-Series). But they also pointed out that the effect of supercharger leakage needs to be borne in mind, particularly at low supercharger rotational speed.

Re-matching a turbocharger to a twin-charged system can also be beneficial for the low-load parasitic losses. Here the parasitic losses refer to the pumping work incurred by the slightly higher backpressure in a turbocharged system. Since a supercharger can be used to provide boost at low engine speed, a larger size of a turbocharger can be selected in a compound charging system. This will not only improve the turbocharger's isentropic efficiency in the high speed and high load region, but more importantly reduce the fuel economy at part load due to a reduction in the pumping losses. For example, King et al. [70] stated that by deploying a larger turbocharger in HyBoost system, an average 2% improvement in BSFC at the key driving cycle engine speeds and loads occurs.

The Miller cycle, which can be realized by either early or late intake valve closing (EIVC or LIVC), is also a viable approach to reduce the low-load parasitic losses (the parasitic losses here refer to the throttling losses) and NOx output in a supercharged gasoline engine [1, 93-99]. Li et al. [93] tested the performance of EIVC and LIVC operation for a twin-charged, high compression ratio, direct-injection gasoline engine. It was demonstrated that at low load (2000 RPM, 4 bar BMEP), the fuel efficiency was better for EIVC than LIVC, primarily due to the reduced throttling losses. The improved fuel-air mixing and enhanced in-cylinder turbulence strength were also noted in an EIVC operation, which was consistent with the research by Bozza et al. [100]

The ‘Five-stroke’ internal combustion engine [97, 101-106], which was proposed and patented by Schmitz, is also a concept to increase the expansion ratio without shortening the compression ratio. The Five-stroke engine is comprised of two outer high-pressure cylinders working in conventional four stroke cycle and one inner low-pressure cylinder working in only expansion and exhaust stroke (see **Figure 24**). By optimizing the geometric compression ratio of the outer cylinders and the displacement of the inner cylinder, the research by Li T et al. [97] showed that this concept can not only improve the fuel conversion efficiency at the middle and high loads, due to the increased combustion efficiency, reduced exhaust energy and the extra expansion work in the inner cylinder, but also facilitate a reduced fuel consumption at low loads, where the reductions in the pumping loss and the exhaust energy are the primary contributions. Compared to the baseline engine, the fuel consumption of the most frequently operated conditions were improved by 9-26%. However a much higher intake boost was anticipated due to the further downsizing. In this context, a two-stage system is currently considered as a potential solution and the novel supercharging technology discussed in this paper might be needed in a compound charging system to provide the additional boost requirement.

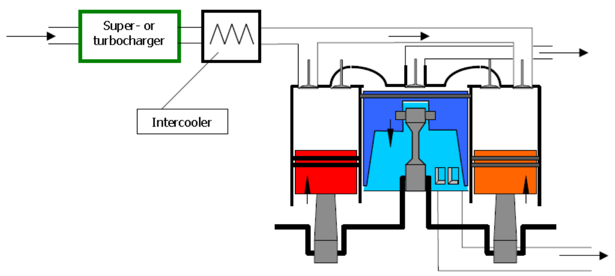

Figure 24. Three cylinder 5-stroke engine. [102]

8 Future development for a mechanically supercharged passenger car engine

Table 2 shows the summary of each assessed technologies from the literature concerning the development of a mechanically supercharged passenger car engine. It can be seen that the separation of the supercharger speed from the engine speed, compressor isentropic and volumetric efficiency improvement and supercharger mechanism innovation seem to be the potential trends for mechanically supercharging a passenger car engine and there is a degree of coupling between them.

Table 2. Summary of each technology from the literature concerning the development of a mechanically supercharged passenger car engine

Future direction / Potential Trends	Low-end torque enhancement	Transient driveability improvement	Low-load parasitic loss reduction
Separation of engine and supercharger speed	CVT mechanism, Pressure wave supercharger, clutch, SuperGen, Lontra and HyBoost	CVT mechanism, SuperGen and HyBoost	CVT mechanism and clutch
Isentropic efficiency improvement	Pressure wave supercharger, TurboClaw, Eaton TVS V-Series, Honeywell novel compressor, SuperGen	TurboClaw, Honeywell novel compressor and Eaton TVS V-Series	
Volumetric efficiency improvement	Eaton TVS V-Series	Eaton TVS V-Series	
Supercharger mechanism innovation	Split cycle engine, Pressure wave supercharger, SuperGen, Lontra and VanDyne SuperTurbo	Pressure wave supercharger, SuperGen and VanDyne SuperTurbo	Positive displacement in low speed and Five stroke engine
Others	Combustion optimization, material innovation	Advanced control, combustion optimization and material innovation	Active bypass, turbocharger/supercharger matching, advanced control and Miller cycle

From the findings in the literature, it seems that neither purely mechanical nor purely electric supercharger system would be able to provide an ideal performance at high load in the near future, due to the nature of a direct connection for a mechanical supercharger with the crankshaft thus constraining its freedom and the fact that a e-boosting system currently is not reliable and durable. A hybrid approach such as SuperGen that features both a mechanical and electric path might be a potential trend that bridges the gap between mechanically and electrically supercharging an engine. In addition, mechanically variable-drive-ratio supercharging system, although its ratio range is not as wide as an electric or a hybrid supercharging system, could also be adopted as a low cost alternative that enhances the performance of an engine in the area of both low-end torque and transient behaviour.

The engine displacement of a supercharged engine should be further optimized in order to satisfy the power requirement of the frequent operating points without the assistance of boosting devices, especially a clutch is fitted. In addition, the low-load parasitic losses incurred by constantly driving a supercharger system (if a clutch is not equipped) is suggested to be either mitigated by adopting a variable drive system or offset via the energy that is reclaimed by the expansion work of a 'supercharger' depending on the ratio range. This approach could also benefit for the transient behaviour as the clutch engagement is not required, thus NVH issues could be largely addressed. Future trends could also include boosting system matching optimization and Miller cycle that is able to offset some (if not all) of fuel efficiency penalty incurred by driving the supercharger from other related technologies. If an automatic transmission gearbox is considered, a systematic approach that takes the trade-off between the part-load fuel efficiency and the transient behaviour is another important area that needs to be noticed and optimized.

For the supercharger compressor itself, the development will be continuous for its isentropic and volumetric efficiency. In addition, the matching of a supercharger with an engine, especially when a two-stage compound charging system is considered, should be evaluated extensively, which must take both devices into consideration.

In order to achieve the competitive functionality, a proper control strategy is always a driving factor. During a transient event, the boost should be triggered from the supercharger seamlessly that behaves like a NA engine and if a two-stage compound charging system is considered, the boost should be seamlessly handed over from the supercharger to the turbocharger after the target is reached. This will require a sophisticated and robust control module and need an extensive calibration in order for the system to operate well for the whole engine operating range. Although a traditional proportional and integral (PI) control could be adopted here but a more sophisticated control algorithm such as neural network is preferred in the future development.

9 Modelling methodologies

Modelling methodologies are being discussed in the present work as a separate section because the fuel efficiency of a boosted vehicle might not be properly simulated using just steady-state engine lookup maps, an approach which is however commonly used in a vehicle-level driving-cycle fuel economy simulation [69].

There are two approaches currently in the literature that utilize steady-state engine lookup maps to simulate the driving cycle fuel economy including engine-level consolidated point methodology and vehicle-level kinematic mode.

Consolidated point methodology: This uses a least squares method to reduce the number of engine speed and load operating points that an engine would operate during a driving cycle, from typically around hundreds to only several points. The point is then given a time weighting based on total driving cycle time [107]. The calculated fuel economy using the engine steady-state lookup tables has been proven to be within 3% error of running full vehicle but with approximately 40 times less computational time [66].

Kinematic mode: This is a backward-looking method directly imposing the driving cycle speed to back-calculate the engine speed and load using the aerodynamic, rolling resistance and road characteristics without considering the transient differences in charge air handling systems [108, 109]. The instantaneous engine speed and load is used for lookups in speed-load dependent engine maps to calculate instantaneous rates of fuel consumption, emissions etc. In addition, since the engine speed and torque are directly calculated these may not reflect the driver's perception of the vehicle performance. However, this method is a relatively simple one and it requires less knowledge of the vehicle control and is still considered to be a valid methodology to simulate and compare naturally-aspirated vehicle performance.

An example is made here to explain why the consolidated point method (termed Minimap methodology in some literature such as [10,49,74,80,92] and kinematic mode cannot capture the boost system transient response and cannot exhibit a real fuel economy in a driving cycle for comparing boosted engines with different transient performance [66]. During a vehicle acceleration event, the transient performance of the engine will influence the driver's pedal demand, which can further cascade to the transmission shifts event. If continuously pressing the pedal still cannot produce the expected torque performance due to the slow boost response, a down shift might be operated, which will result in the engine running at a higher-speed and lower-torque region where the efficiency drops. Thus a proper numerical simulation should incorporate a dynamic driver model to control the engine parameters in a forward-direction manner. However, most of the vehicle simulation currently conducted does not do this.

Half-dynamic mode: Most driving-cycle fuel economy simulation had been conducted kinematically due to the fact the dynamic solution was extremely time-consuming and a robust solution was only available for kinematic simulation. Typically, a PI controller was used to target vehicle speed in a driving cycle, but this required laborious tuning of different gains for a particular vehicle over a particular driving cycle to avoid control noise. Recent trend in vehicle simulation is adopting a model-based feed-forward dynamic solution with corrective feedback loop (PI gains) to accurately follow driving-cycle speed targets [110]. Due to the fact that the PI controller of a dynamic solution will only be used to correct small errors that exist, the simulation speed is significantly increased. Note that a fully developed dynamic model should incorporate dynamic modules for both the vehicle and driveline. Currently the most adopted approach for vehicle simulation only utilizes the dynamic mode for the vehicle without a detailed dynamic analysis for the engine. This method as opposed to the kinematic mode mentioned above and the full-dynamic mode below is termed half-dynamic mode, which features a more-developed vehicle control model to determine the necessary accelerator and brake pedal position in order to follow the required driving cycle target but only with engine steady-state lookup tables.

Full-dynamic mode: Unlike the two approaches discussed above, a full-dynamic mode refers to those with a detailed engine model and a more-developed driver control unit. By doing this the dynamic response of the engine can be captured and the driver behaviour can also be simulated. Note that this methodology is extremely central processing unit (CPU) intensive when trying to simulate a drive cycle. A fast running model (FRM) might need to be used in this case to save some simulation time [110]. This is, in essence, achieved by lumping pipe objects together on the intake and exhaust side of a detailed engine model. Although this method cannot capture higher frequency wave dynamic content, a high level of fidelity of the low frequency and primary engine order dynamics is maintained, which is normally of interest to transient simulations.

This mode allows the development of the physical control system for both the vehicle and the engine [108]. In addition, the down-speeding and transmission shift frequency effect on fuel efficiency might only be able to simulate in this mode for different boosting solutions [69], which opposed to the conventional engine-map based vehicle models has the capability to model transient phenomena such as turbo-lag during drive away and gear shifts [110].

10 Conclusion

Mechanically supercharging a passenger car engine is considered to be an alternative or complementary approach to enable heavy downsizing. Since it is directly driven by the engine crankshaft, the transient performance of an engine boosting system realized either solely by a supercharger or by a combination of supercharger and turbocharger is significantly better than that of its turbocharged counterpart. The following conclusion are drawn from reviewing the literature:

1: There are two main types of compressor that are currently adopted in passenger car engines including positive-displacement and centrifugal compressors.

2: Some of the current state-of-the-art technologies for mass production downsized supercharged engines indicate some synergies between the engine and the boosting machine: a strong relationship between downsizing, driveability and down-speeding.

3: Low-end torque enhancement, transient driveability improvement and low load parasitic loss reduction are the three development directions for a supercharger system, among which the adoption of a CVT to decouple the supercharger speed from the engine speed, compressor isentropic and volumetric efficiency improvement and supercharger mechanism innovation seem to be the potential trends for mechanically supercharging a passenger car engine.

4: The method to model a supercharged-based vehicle is of importance to evaluate the accurate fuel economy of a passenger car engine in a real driving cycle. It is demonstrated that both the consolidated point methodology and the kinematic model is not able to capture the transient response of a boosted engine and the driver behaviour. A half-dynamic model improves the model performance by taking driver perception into consideration. To fully understand the performance of a boosted vehicle, a fully-dynamic model is suggested and a fast funning model might need to be built depending on the development time frame.

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Notation

T	torque
I	inertia
$\dot{\omega}$	Angular acceleration

Abbreviations

ARTEMIS	assessment and reliability of transport emission models and inventory systems
BMEP	brake mean effective pressure
BSFC	brake specific fuel consumption
CSC	China Scholarship Council
CVT	continuously variable transmission
DEP	divided exhaust period
DF	downsizing factor
EGR	exhaust gas recirculation
EIVC	early intake valve closing
FEAD	front-end accessory drive
FGT	fixed geometry turbine
JLR	Jaguar Land Rover
LIVC	late intake valve closing
JLR	Jaguar Land Rover
NEDC	new European driving cycle
NVH	noise vibration and harshness
OEMs	original equipment manufacturers
PI	proportional and integral
PR	pressure ratio
PWS	pressure wave supercharger
RGF	residual gas fraction
RPM	revolutions per minute
SC	supercharger
SS	steady state
TFSI	turbocharged fuel stratified injection
TSI	twin-charged stratified injection
VGT	variable geometry turbocharger
w/EGR	without exhaust gas recirculation

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